1968

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http://hdl.handle.net/10945/11962

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by

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Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

NAVAL POSTGRADUATE SCHOOL
June 1968

ABSTRACT

An experimental stainless steel heat pipe using water as the working fluid and 400 mesh stainless steel screen for a wick was designed and tested to determine the effect of gravity and nucleate boiling on heat pipe performance. The results of heat pipe operation at various angles of inclination in a gravity field are presented and compared with the existing theoretical predictions.

The maximum heat flux obtained experimentally at angles of inclination less than 90 degrees was less than the predicted value by a factor of two or three. The maximum heat flux obtained for an angle of inclination greater than 90 degrees was much higher than that predicted. In addition, nucleate boiling noise was detected during operation at angles of inclination greater than 90 degrees.

The experimental results coupled with visual examination of the pipe after operation indicate that the pipe was not performing satisfactorily. Recommendations for a better design of an experimental heat pipe are presented.

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NOMENCLATURE

A	Area, ft^2
b	Screen permeability factor
g	Acceleration due to gravity, ft/sec^2
L	Latent heat of vaporization, BTU/lb
$\dot{m}$	Mass rate of flow, lb/sec
Q	Heat flow, watts, BTU/sec, BTU/hr
R	Radius of vapor passage, ft
R_r	Reynolds number = $Q/\pi LR\eta_v$
R_w	Outside radius of wick structure, ft
r_c	Effective capillary radius of wick, ft
Z	Heat pipe length, ft
γ	Surface tension, lb/ft
δ	Incremental error
ϵ	Wick void fraction
η_v	Dynamic viscosity of the vapor
θ	Contact angle
ν	Kinematic viscosity
ρ_l	Density of liquid, lb/ft^3
ρ_v	Density of vapor, lb/ft^3
ϕ	Angle of inclination of heat pipe, measured from the vertical with evaporator above condenser

ACKNOWLEDGEMENTS

The author would like to extend his thanks to Dr. P. J. Marto of the Naval Postgraduate School for his many constructive criticisms of this thesis and to Professor M. D. Kelleher for his comments in reading the rough copy of the thesis as departmental reader. Also Mr. Frank Abbe of the Machine Facility is due many thanks for putting in many long hours and making many practical suggestions that aided in the construction of the apparatus.

Many thanks also go to my wife, Donna, who typed the original rough copy and who exhibited exemplary patience during this endeavor.

I. INTRODUCTION

Background

The transportation of large amounts of heat plays an increasingly important role in present day engineering problems and applications. It is not surprising that a major effort has been devoted to research to find more efficient ways to transfer large amounts of heat. The heat pipe is one such device that has recently been developed which is capable of transferring large amounts of heat efficiently. Copper, which is usually regarded as one of the best conductors, is not nearly as efficient as the heat pipe. For instance, if thermal power of 10,000 watts were applied to one end of a solid copper bar one inch in diameter and one foot long, the temperature difference along the bar could theoretically exceed 30,000°F. This same amount of thermal power could be transferred through a heat pipe of similar dimensions with only a small temperature difference [1]. In this device, heat is transferred by the evaporation of a liquid in one section of the pipe and the condensation of the vapor at a colder section of the pipe. An ordinary reflux capsule (or thermosiphon) transfers heat in this manner, also. The unique property of the heat pipe is that no outside source of energy is required to return the condensate to the evaporator. In a heat pipe, the return of the condensate is accomplished by the use of porous wicking material (such as wire screen, small channels, or sintered porous material) inside the pipe. The heat pipe is a self-contained, closed system requiring no outside source of energy other than the energy being transported.

The basic principle of the heat pipe was first discovered by R. S. Gaugler in 1942 but was not utilized until 1963, when G. M. Grover independently discovered the device and named it a "heat pipe" [1].

Since 1963, Grover [2], Cotter [3,4], and others have done research, both theoretically and experimentally, to determine design and operating characteristics of these devices.

Theory of Operation

The heat pipe is essentially a closed, hollow structure whose inside wall is covered by a porous wick. It is not necessary that the wick be placed against the inside surface of the container, although this will normally be the best place for it. Condensation and evaporation take place at the liquid-vapor interface, therefore if the wick is placed against the inside surface the radial transfer of heat necessary for operation will occur through the medium of highest thermal conductance and radial temperature differences will be minimized. If for some reason it is not desirable to place the wick next to the wall, it is possible to use fins to conduct the heat to the wick and thus reduce the radial temperature difference.

For theoretical simplicity, assume the heat pipe is a long, hollow, closed cylinder with the wick adjacent to the inside surface. (Figure 1). The theory for steady state operation for such a pipe has been developed by Cotter [3] and is briefly stated below.

The assumptions made in Cotter's theory are:

(1) The pipe has a large length to diameter ratio.

(2) The pipe contains sufficient pure liquid to fill the wick.

(3) The vapor space contains no non-condensable gases.

(4) The vapor flow is in the incompressible regime and inertial forces predominate.

(5) The pressure distribution in the wick obeys the hydrostatic laws for an incompressible fluid.

(6) Flow through the wick may be given by a form of Darcy's law

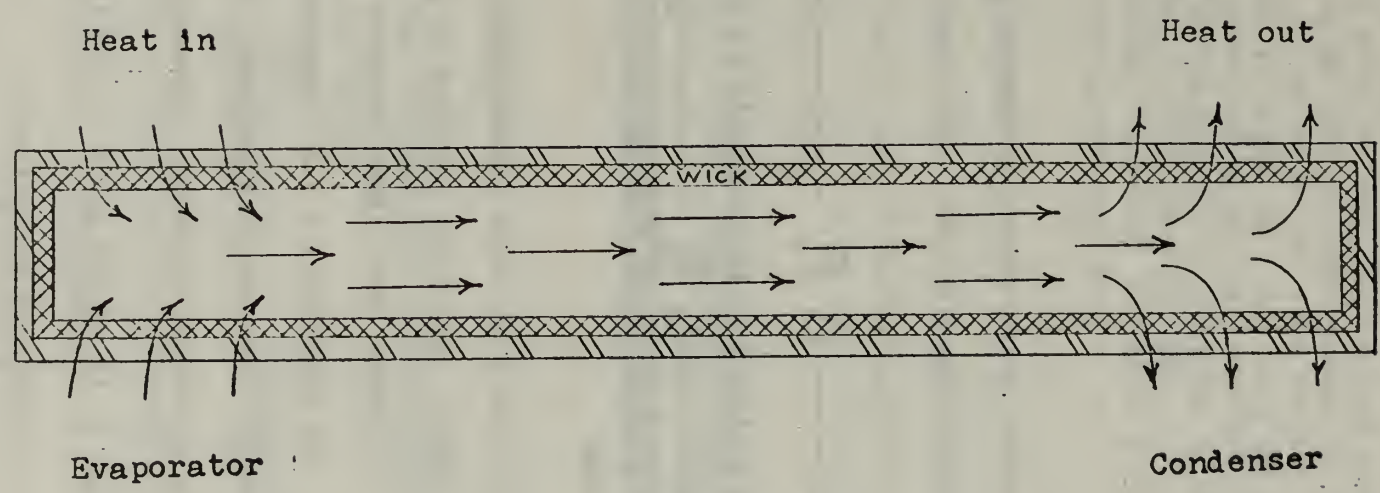


Figure 1. Schematic Drawing of the Heat Pipe

for flow through porous media.

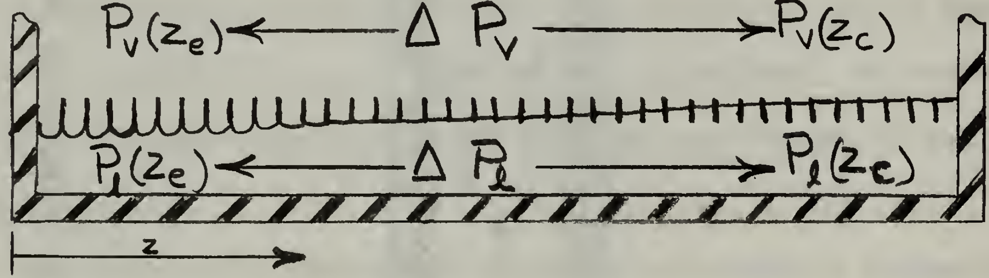
(7) The diameter of the pipe is constant.

(8) Heat addition and removal is uniform.

(9) Nucleate boiling does not occur in the wick.

Axial heat transfer is accomplished by flow of the vapor from the hot end of the pipe to the cold end. The vapor is condensed at the cold end of the pipe and is returned to the hot end by means of the wicking material. The driving force for successful operation of the pipe is capillary action.

In order to arrive at the equations governing the operation of the heat pipe, consider the sketch below. The sketch represents a partial



cross-section of the heat pipe which schematically shows the wick and part of the vapor cavity. The vapor pressure at a given axial position is given by $P_v(z)$. Likewise, the liquid pressure at some axial position is given by $P_l(z)$. The subscripts c and e represent positions in the condenser and evaporator sections, respectively. Writing a pressure balance for the pipe gives the following expression:

$$(P_v(z_e)-P_v(z_c)) + (P_v(z_c)-P_l(z_c)) + (P_l(z_c)-P_l(z_e)) + (P_l(z_e)-P_v(z_e)) = 0$$

Now at any axial position, z, the interface between the liquid in the wick and the adjacent vapor must assume a local radius of curvature, r(z), such

that the surface tenxion, γ , supports the difference in pressure between the liquid and the vapor. Thus,

$$P_v(z)-P_l(z) = 2\,\gamma \cos\theta/r(z)$$

where θ is the contact angle between the surface of the liquid and the capillary structure. In the condenser, the radius of curvature, $r(z_c)$, will be large due to condensation and may approach infinity while the contact angle θ approaches 90 degrees so that

$$P_v(z_c)-P_l(z_c) = 2\,\gamma \cos\theta/r(z_c) \approx 0$$

On the contrary,in the evaporator section, the radius of curvature, $r(z_e)$, will be small due to evaporation and θ approaches zero so that the pressure drop across the liquid-vapor interface may be written

$$P_v(z_e)-P_l(z_e) = 2\,\gamma\,/r(z_e)$$

Letting $P_v(z_e)-P_v(z_c) = \Delta P_v$ and $P_l(z_c)-P_l(z_e) = \Delta P_l$, with the above simplifications, then the pressure balance may now be rewritten as

$$\Delta P_v + \Delta P_l = 2\,\gamma\,/r(z_e) \qquad (1)$$

Darcy's law for flow through porous media yields an expression for the pressure drop in the liquid

$$\Delta P_l = \frac{b\nu ZQ}{2\,\pi\,(R_w^2-R^2)\,\epsilon\, r_c^2 L} + \frac{\rho_l\, g\, z \cos\phi}{g_o} \qquad (2)$$

where b is a dimensionless constant which depends on the geometry of the porous wick, ν is the kinematic viscosity, Z is the length of the pipe, Q is the heat applied, R is the radius of the vapor cavity, and R_w is the outer radius of the wick. The wick void fraction is represented by ϵ and r_c is the effective capillary radius. L is the latent heat of vaporization of the fluid, ρ_l is the density of the liquid, g/g_o is the gravitation

term and ϕ is the angle of inclination of the pipe with the vertical.

The effective capillary radius, r_c, for a wire screen or porous wicking material must be determined by a capillary rise experiment in which the height of liquid supported by the wicking material is experimentally measured. This height h, is used in the relationship

$$r_c = 2\gamma\cos\theta / \rho(g/g_o)h$$

to define the effective capillary radius. In the above relationships, ρ is the density of the liquid used in the experiment, and $\cos\theta$ is assumed to be unity. If the capillary structure is a small tube of circular cross-section, then the effective capillary radius turns out to be the inner radius of the tube as would be expected.

Two major regimes exist for flow in the vapor phase. These regimes depend on the magnitude of a Reynolds number, R_r, based on the radial flow velocity of the vapor at the channel wall, the vapor cavity radius, R, the vapor density, ρ_v, and dynamic viscosity, η_v. The previously assumed conditions for vapor flow yield the following relationship for the pressure drop in the vapor phase:

$$\Delta P_v = \begin{cases} \dfrac{4\,\eta_v Z Q}{\pi \rho_v R^4 L} , & R_r \ll 1 \\[2ex] \dfrac{(1-4/\pi^2)\,Q^2}{8\,\rho_v\,R^4 L^2} , & R_r \gg 1 \end{cases} \tag{3}$$

The above equations represent the steady state solutions for the pressure drops in the liquid and vapor phases. In general, the Reynolds number, R_r, will be greater than 1.

The local radius of curvature, r(z), will be at least as large as the vapor cavity radius, R, and may be as small as the effective capillary radius of the wick, r_c. The effective capillary radius determines the maximum pressure difference which may exist between the liquid and the

vapor. If r(z) is the effective capillary radius, r_c, equation (1) may be rewritten as

$$\Delta P_v + \Delta P_l = 2\gamma / r_c \tag{4}$$

Substituting equations (2) and (3) into (4) yields

$$\frac{(1-4/\pi^2)\, Q^2}{8\rho_v R^4 L^2} + \frac{b \nu Z Q}{2\pi (R_w^2 - R^2)\, \epsilon\, r_c^2 L} + \frac{\rho_l\, g\, Z \cos \phi}{g_o} = \frac{2\gamma}{r_c} \tag{5}$$

The solution of equation (5) for Q gives the maximum heat flux that can be attained in a heat pipe which is limited by capillary action only.

Physical indications of having reached the maximum heat transfer limit are a sudden rise in temperature in the evaporator and a sudden drop in temperature in the condenser.

It can be seen from equation (5) that a number of factors govern the operation of a heat pipe. These factors may be grouped into three broad categories: the maximum heat flux, the pipe geometry and size, and the properties of the working fluid. By specifying the proposed application and selecting a working fluid for the temperature range desired, one can design a heat pipe of size and geometry to fit the application. In other words, the application of the pipe and the working fluid determine the design of the pipe.

In deriving the equations for the operation of a heat pipe, the assumption was made that nucleate boiling did not occur. The reason for the above assumption was because it was presumed that the formation of bubbles in the wick would prevent the condensate from returning to the evaporator or would at least impede flow to the evaporator. The result of insufficient flow to the evaporator would be the formation of hot spots and premature drying out of the evaporator.

Previous Investigations

Since the work by Grover in 1963, much work has been done investigating the properties and uses of the heat pipe. In 1965, Grover, et. al. [2] investigated the use of a heat pipe as a heat removal device for thermionic devices in space. B. D. Marcus [5], apparently working independently at TRW Space Technology, arrived at a theory to determine the maximum heat flux for a heat pipe. Marcus arrived at a method which optimized the thickness of the wick such that the wick was just large enough to permit the condensate flow rate corresponding to the maximum heat transfer rate and just small enough to avoid nucleate boiling in the wick at the maximum heat transfer rate. Recently Cotter [4] investigated the transient modes of heat pipe operation. He discussed flow properties during startup and the mechanisms by which startup can fail. Two phenomena can cause startup to fail. The first one being the attainment of the superheating limit at which hot spots form in the evaporator excluding the liquid from where it is needed due to boiling in the capillary structure. The second includes all other mechanisms of failure which result in insufficient return of condensate to the evaporator. J. E. Kemme [6] ran a series of experiments with a horizontal heat pipe (thus eliminating the gravity effect) using liquid sodium and potassium in heat pipes which had wicks composed of rectangular channels. Kemme in some cases covered the channels with fine mesh screen and studied the effect of the screen on the performance. Ten heat pipes constructed of nickel were utilized. These tests demonstrated that the heat transfer characteristics of heat pipes with fixed outside dimensions can be changed significantly by variations in the wick structure. The practical temperature range over which such systems operate will be determined by the working fluid. Kemme [7] also investigated the entrainment limit and found that this phenomenon could limit heat transfer.

The entrainment limit occurs when the velocity in the vapor phase becomes great enough to entrain droplets of liquid from the wick thus preventing the condensate from returning to the evaporator. Carnesale, et. al., [8] developed a theory of operation for heat pipes and experimentally investigated pipes which utilized water for the working fluid and packed monel beads for the wick. Carnesale also investigated the effect of the inclination of the pipe with respect to gravity in his experiments. Carnesale's group developed their own theoretical equations which are similar to equation (5). Their experimental results were in close agreement with their theoretical results. J. E. Deverall and E. W. Salmi [9] have tested a heat pipe in space and have found that the pipe operated normally. In addition, Deverall and Salmi ran an experiment by attaching a heat pipe to a vibration tester and operating the pipe as the vibration tester was in operation. They found that the vibrations had no adverse effect on operation and in some cases performance was improved because the vibration made the wick wet more rapidly and completely. D. M. Ernst [10] has performed further theoretical evaluation of Cotter's work [3] and has come up with some interesting modifications to Cotter's equation for maximum heat flux (equation (5)). Ernst modified the boundary condition of the condenser and arrived at the following equation for the pressure drop in the vapor.

$$\Delta P_v(z) = Q^2/8 \rho_v R^4 L^2 \qquad (6)$$

Samuel Katzoff [11] investigated the use of heat pipes and vapor chambers for thermal control in spacecraft. Katzoff discussed some of the practical considerations concerning the use of heat pipes in space applications. Some of these considerations were wicking materials, working fluids, and the effect of contaminants such as non-condensable gases. The working

fluids which he investigated were alcohol, water, and a two-component fluid consisting of ethanol and methanol.

<u>Thesis Objectives</u>

The objectives of this thesis were to design and operate a heat pipe, and to determine what effect gravity and nucleate boiling in the wick have on heat pipe performance.

The first objective was to determine the operational details of manufacturing a laboratory heat pipe for experimental use. Examples of the operational details sought were: (1) What difficulties were encountered in the manufacture of the pipe? (2) Was the cleaning procedure correct? (3) Was the instrumentation adequate? (4) What factors should one keep in mind when designing a heat pipe?

The effect of gravity appears only in the liquid pressure drop term in equation (5). The second objective was to determine if the effect of gravity on the liquid pressure drop is accurately represented and if gravity effects the operation of the heat pipe in any way not predicted by Cotter's theory.

Cotter [3], Kemme [6], and others have stated that the formation of bubbles in the wick in the evaporator section of the pipe will limit the maximum heat flux that the pipe will transfer. In other words, the heat transfer mechanism of the pipe is evaporation and heat pipe operation fails when boiling occurs in the wick. Consequently, the final objective was to determine if boiling occurred during operation or at least at maximum heat flux and to determine the effect of such boiling on the operation of the pipe.

II. EXPERIMENTAL EQUIPMENT AND PROCEDURES

Equipment Design and Construction

Once the application of a heat pipe has been decided upon, a suitable working fluid must be chosen. The working fluid must be capable of wetting the heat pipe material. It must also have high surface tension, high latent heat of vaporization and heat capacity, low viscosity, and a boiling point compatible with the operating temperature of the proposed heat pipe. In addition, the working fluid should, if possible, be readily available, easy to handle, and inexpensive. Water was selected for this application because it met most of the above criteria and the temperature range for this experiment was to be between 200°F and 300°F. Other liquids that have been used are sodium, potassium, and alcohol [6], [8], [11].

The heat pipe material should have high thermal conductivity and be compatible with the working fluid selected. Stainless steel, if properly cleaned, has good wetting characteristics with water. Stainless steel is difficult to machine because of its hardness. When stainless steel is used with water as a working fluid, great care must be taken to ensure that the steel is perfectly clean.

The initial design called for a wick consisting of rectangular grooves or slots cut into the stainless steel pipe wall and covered with a fine mesh stainless steel screen. The effective capillary radius of the screen determines the maximum driving force for flow in the wick. The driving force is inversely proportional to the effective capillary radius, therefore it is desirable to have the effective capillary radius as small as possible. For porous media, the pressure drop in the liquid is inversely proportional to the square of the capillary radius. It can be seen that in

trying to increase the driving force in the wick by decreasing the capillary radius it is possible to create a pressure drop in the liquid large enough to inhibit operation. This situation may be avoided by the use of covered channels. The driving force is still provided by the small capillary radius of the screen but the pressure drop is governed by the size and shape of the channels. Since the pipe was to be required to perform against gravity, a wire mesh having a very small effective capillary radius was required to cover the channels. Very little information concerning the effective capillary radius of wire mesh was available. Katzoff [11] had run capillary rise tests on six wire mesh screens and from his data it appeared that a 300-400 mesh screen would lift a column of water about 16-19 inches.

The final design for the heat pipe consisted of a 1.5 inch outside diameter stainless steel pipe having a wall thickness of 0.18 inches and a wicking material consisting of three layers of 400 mesh stainless steel screen. The wicking material was held in place against the inside of the pipe wall by a helical wire spring inserted inside the pipe. The large wall thickness was required to allow thermocouple wells to be drilled axially into the wall of the evaporator section.

A series of tests were run to determine the effective capillary radius of the 400 mesh stainless steel screen selected and the height of a column of water that could be supported by the mesh. The screen selected was able to support a column of water 21 inches high.

One of the theoretical assumptions was a large length to diameter ratio for the pipe. The length of the pipe had to be limited so that the pipe would operate in a vertical position with the evaporator uppermost, i.e., the length of the pipe could not be greater than the height of the column of water that the screen would support. In fact, it should be less. On

this basis, a length of twelve inches and a diameter of about one inch were selected. The pipe was cut 12.36 inches long to allow the 0.18 inch thick end caps to fit in each end. The resulting inside length of the pipe, including the wick, was twelve inches. One end cap was fitted with a ¼ inch stainless steel tube and vacuum valve to use for filling the pipe and was then welded in place.

The reason for the use of the mesh wicking material rather than the rectangular grooves was because of the difficulty encountered in machining the grooves in a stainless steel pipe. The grooves in the wick were to be cut 0.010 x 0.060 inches in a flat piece of stainless steel 0.125 inches thick. The steel sheet was then to be formed into a cylindrical pipe by rolling and seam welding. There was no facility available locally that could roll the steel without damaging the grooves, so some consideration was given to designing a pipe of square cross-section. This scheme was abandoned because of difficulties in machining the small grooves in the stainless steel. Some of the difficulties encountered were a high rate of blade breakage and mechanical inaccuracies exemplified by slight vibrations of the milling machine which caused the blades to cut through or bend the metal between the grooves. The above-mentioned difficulties caused the construction of the pipe to fall behind schedule.

The final design created a large pressure drop in the liquid phase and resulted in the maximum theoretical heat transfer being reduced by a factor of ten or from about 6800 BTU/hr to 680 BTU/hr. In order to measure the pressure in the liquid phase a series of six pressure taps were drilled into the pipe. Three taps were placed at each end of the pipe and spaced 0.7, 1.7, and 2.7 inches from the ends, respectively.

The requirement to measure the temperature distribution in the pipe

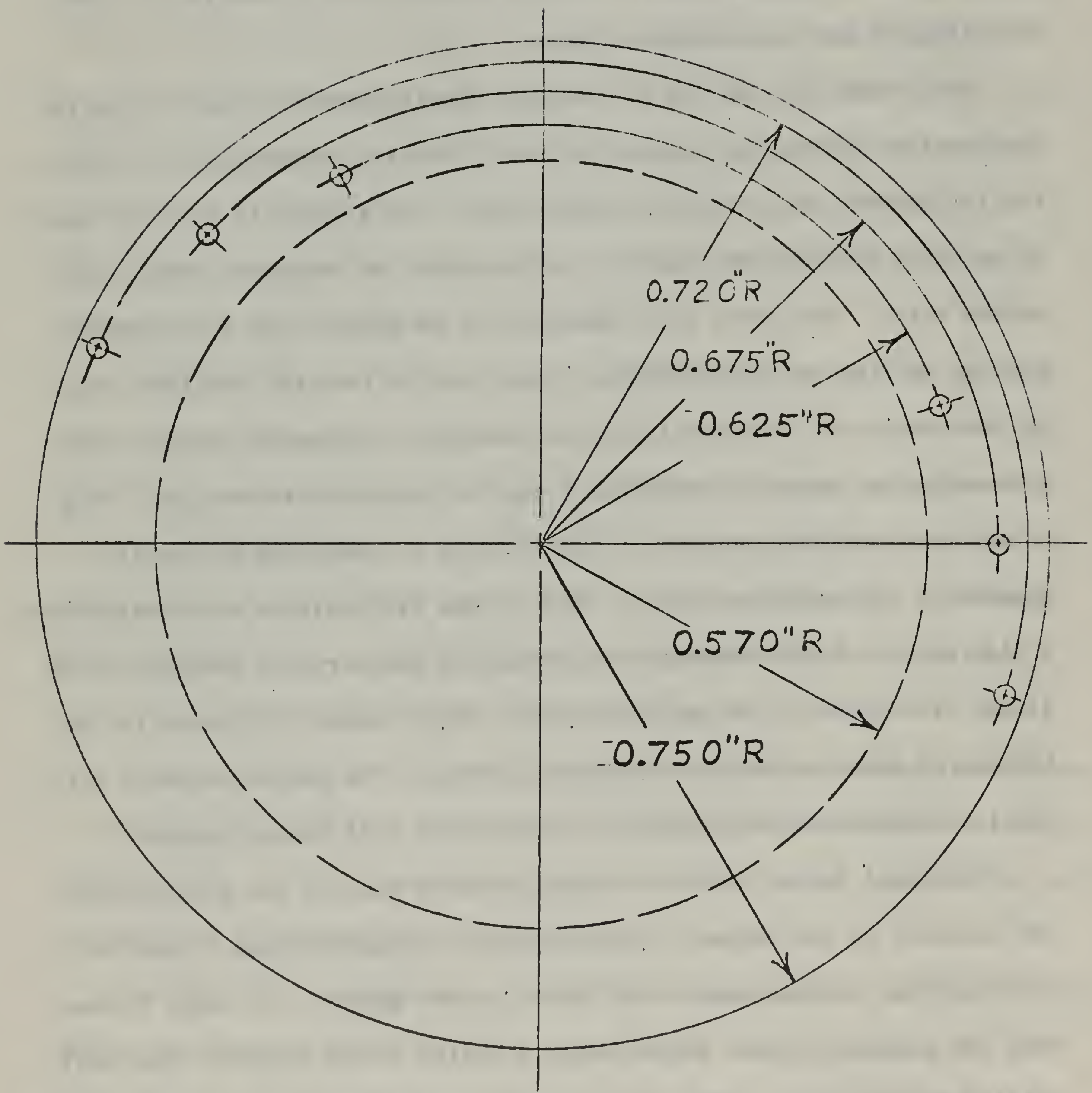


Figure 2. Thermocouple Positions in Evaporator Wall

wall necessitated placing thermocouples in the wall of the evaporator. Six holes 0.040 inches in diameter were drilled an axial distance of one inch into the pipe wall. The location of these holes is shown in Figure 2. Much difficulty was encountered in drilling the holes for the thermocouples. Countless drills were broken and the drills that did not break had a tendency to bend. In one instance the drill penetrated the interior of the pipe and the hole had to be welded shut. In two other cases the drills came out of the outer surface of the pipe and these holes had to be closed also.

After the pressure taps and wicking material had been installed in the pipe and prior to sealing the remaining open end, the pipe was first thoroughly degreased with acetone and dried by a heat lamp. After drying, the pipe was bright dipped ten to fifteen minutes in an aqueous solution of 12% (by volume) nitric acid at 160°F. The pipe was then rinsed three times in distilled water and the remaining end cap was welded into place and the holes for the thermocouples were drilled. The heat pipe was then heated to about 500°F and outgassed with a vacuum pump for twelve hours. At this point it was discovered that the vacuum pump in operation was faulty and oil was getting into the heat pipe. The cleaning process was then repeated with some modification. Since the pipe had been sealed and the only access to its interior was through the filling valve and the pressure taps which were covered on the inside by the wick, the acetone, acid, and distilled water each had to be forced through the pipe. The pipe was then heated and outgassed again using a different vacuum pump.

A cooling coil of 3/8 inch copper tubing was then tightly wrapped and soldered to one end of the pipe to form a water jacket for cooling the condenser. The length of the condenser section was 4-3/4 inches.

The heating element for the pipe was THERMOCOAX wire and SAUEREISEN electrical cement. The heater wire consisted of an inner conductor of nickel-chromium alloy covered with an electrical insulating layer of compressed magnesium oxide. The layer of magnesium oxide was covered with a sheath of inconel. The wire was tightly wrapped around the evaporator section and then was covered with the electrical cement. The purpose of the electrical cement was to hold the heater wire in place and to provide a more uniform heat distribution over the surface of the evaporator.

The heat pipe was filled with water prior to the installation of the heating element. The pipe was filled by distilling 7.5 ml of water from a flask and condensing the vapor in the heat pipe. This amount of water was calculated to be sufficient to completely fill the wick and to have a slight excess of water remaining. With the heat pipe filled and ready to operate, the instrumentation was installed prior to operation. The thermocouples were installed in the pipe wall and along the side of the pipe. The thermocouples used were 40 mil copper-constantan thermocouples referenced against an ice junction. Manometers were connected to the pressure taps to measure the pressure drop in the wick. Copper-constantan thermocouples were also installed at the entrance and exit of the condenser coil to measure the temperature change of the cooling water.

A seven inch length of ½ inch round stock was attached to the pipe so that the pipe could be mounted and rotated in a vertical plane about the horizontal axis of the arm. The pipe was then mounted and wrapped with asbestos until a layer of insulation about 3/4 inch thick covered the pipe completely except for the condenser coil.

Power leads were then attached to a variac which was connected to a 115 volt alternating current power supply. An a-c wattmeter and ammeter were installed between the variac and the heater element to measure the

power supplied and to monitor the current. The requirement to monitor the current was necessary because the heater element wire would only carry seventeen amperes without melting.

Operating Procedures

The first operation of the pipe was in the vertical position with the evaporator at the lower end. Power was supplied at the rate of fifty watts for about fifteen minutes until condensation drops began to form in the tubes leading to the manometers. Further operation caused the pressure in the pipe to build up and to force the water from the pipe into the tubes to the manometers resulting in the pipe drying out. Consequently, at this point operation was stopped. The pipe was then emptied and refilled with 25 ml of water. This amount of water was sufficient to fill the wick and to fill the vapor chamber to a height of one and one-half inches from the bottom with the pipe in a vertical position. Five of the six pressure taps were then sealed and the remaining tap on the condenser end of the pipe was connected to a mercury manometer so that the pressure inside the pipe could be measured. The pipe was then returned to the previous vertical position with the evaporator in the lowest position and heat was applied in increments of fifty watts over a range from zero to seven hundred watts. At each increment the heat input was held constant until the temperatures in the pipe wall stabilized and a series of readings were taken of all temperatures and the internal pressure of the pipe. Noise inside the pipe was monitored with a stethoscope and nucleation could be heard at a heat input of fifty watts and above. The pipe was then rotated to the horizontal position and heat was again applied in various intervals from fifty to four hundred watts. At each interval the temperatures were allowed to stabilize and a series of readings were taken.

The temperature did not appear to stabilize at a power input of four hundred watts and this was the final data point taken. The value of maximum heat flux was determined by plotting the axial temperature distribution and observing the rate of heat input at which the temperatures do not stabilize. The maximum heat flux was taken as the maximum heat input at which thermal equilibrium was attained. The pipe was then rotated to a position 60 degrees from the vertical with the evaporator higher than the condenser and the power was increased in ten watt intervals beginning with twenty watts until the limit was reached. The pipe was then rotated to positions 45 and 0 degrees from the vertical with the evaporator uppermost and the process repeated in each position until the point of maximum heat flux had been reached. Noise level was also monitored at each position of the pipe, but no nucleation or boiling noises could be detected in the 60, 45, or 0 degree positions. Boiling could be heard in the horizontal position.

The rate of flow of cooling water through the condenser was measured at each heat pipe position by means of a stop watch and a weighing tank.

It was noticed that the heat pipe seemed to have a small leak. This was evidenced by the fact that the pipe would lose part of its vacuum if left overnight or longer. The pressure in the pipe at room temperature should be the pressure of saturated water vapor at that given temperature and should be lower than atmospheric pressure. When the pipe had cooled from a series of runs, the pressure would be the saturation pressure at the temperature of the pipe. The pressure would increase overnight, indicating that air had leaked into the pipe. To overcome this problem, or at least to get a common starting point for each series of runs, the pipe was operated in the vertical position, evaporator at the lower end, at a heat input of fifty watts for fifteen minutes with the

loading valve open.

The reason for removing air from the pipe was because non-condensable gases collect in the condenser during operation and reduce the volume of the condenser that the vapor can enter. Consequently, the wick area available for condensation is reduced because the vapor cannot reach all of the wick in the condenser due to the non-condensable gases present.

Measurement Techniques

In attempting to adequately determine the performance of an experimental heat pipe, the parameters to be monitored are the heat fluxes into and out of the pipe, the axial temperature distribution, and the pressure distribution. It is sometimes convenient to be able to watch or listen to the action inside the pipe during operation.

The heat fluxes into and out of the pipe were measured in two different manners. The heat into the pipe was determined by measuring the temperature at various points in the pipe wall and using the resulting temperature distribution in the wall and the equations for steady-state conduction assuming one-dimensional radial heat transfer to determine the heat flux. The heat out of the pipe was determined from the difference in temperature and the mass rate of flow of the cooling water through the condenser.

The axial temperature distribution was monitored by placing two thermocouples along the section of the heat pipe between the evaporator and the condenser. One thermocouple was placed on the outside wall of the pipe two inches from the evaporator section and the second thermocouple was placed on the outside pipe wall four inches from the evaporator section and about ¼ inch from the condenser.

The pipe was not constructed to accommodate visual observation, however it was possible to monitor the noise generated during operation by use of a stethoscope.

III. DISCUSSION OF RESULTS

Theoretical Calculations

The theoretical results were obtained by solving equation (5) for the heat flux. The properties of the fluid were taken to be the properties of water at 212°F and one atmosphere of pressure. Pipe and wick dimensions were the actual dimensions of the pipe and wick. The contact angle, θ, was assumed to be close to zero so that cos θ is approximately equal to one. The wick void fraction was approximated by dividing the difference in weight of a sample of the wire mesh and the weight of a solid piece of stainless steel the same size by the weight of the solid piece of stainless steel. The screen permeability factor, b, was assumed to be twenty.

Equation (5) is of quadratic form which may be readily solved for Q giving:

$$Q = \frac{-B \pm \sqrt{B^2-4AC}}{2\,A} \qquad (7)$$

Where A is the coefficient of Q^2, B is the coefficient of Q, and C is composed of the remaining terms. As it turns out, very small values of the effective capillary radius, r_c, tend to make B so large that the equation yeilds either a very large negative value for Q or a trival solution. Neither of these solutions is meaningful. To avoid this situation, equation (5) may be solved for maximum heat flow by assuming a value for Q and solving the equation by some iterative technique. Figure 3 shows the values of Q versus Ø. The theoretical values of Q were obtained by solving equation (5) by the Newton-Raphson method. A sample calculation is given in Appendix I.

Experimental Results

Most of the experimental results are shown in Figures 3 through 8. Figure 3 shows the theoretical and experimental values of Q versus angle of inclination, Ø. Figures 4 through 8 show the temperature distribution along the length of the pipe as a function of heat input at various angles of inclination, Ø.

The heat pipe did not operate as the theory predicted. For angles of inclination less than 90 degrees, the experimental results were much lower than the theoretical values. There are three possible reasons for this behavior. First, the pressure drop in the wick was much greater than anticipated or second, the working fluid was not wetting the wicking material so that capillary action was not taking place. Either of the two above phenomena would be detrimental to operation resulting in insufficient condensate flow to the evaporator for sustained operation except at much lower heat fluxes than those predicted. The remaining reason for the large discrepancy between theoretical and experimental values for maximum heat flux could also be attributed to not knowing the exact positions of the thermocouples in the heat pipe wall. The error in Q that is introduced by small inaccuracies in the thermocouple positions can be on the order of 100% (Appendix II).

At an angle of inclination of 90 degrees and greater, the device was acting as a reflux boiler, causing the maximum heat flux to be much greater than heat pipe theory would predict.

Failure to operate close to the theoretical predictions can be attributed to manufacturing and cleaning errors which were not apparent until during or after operation.

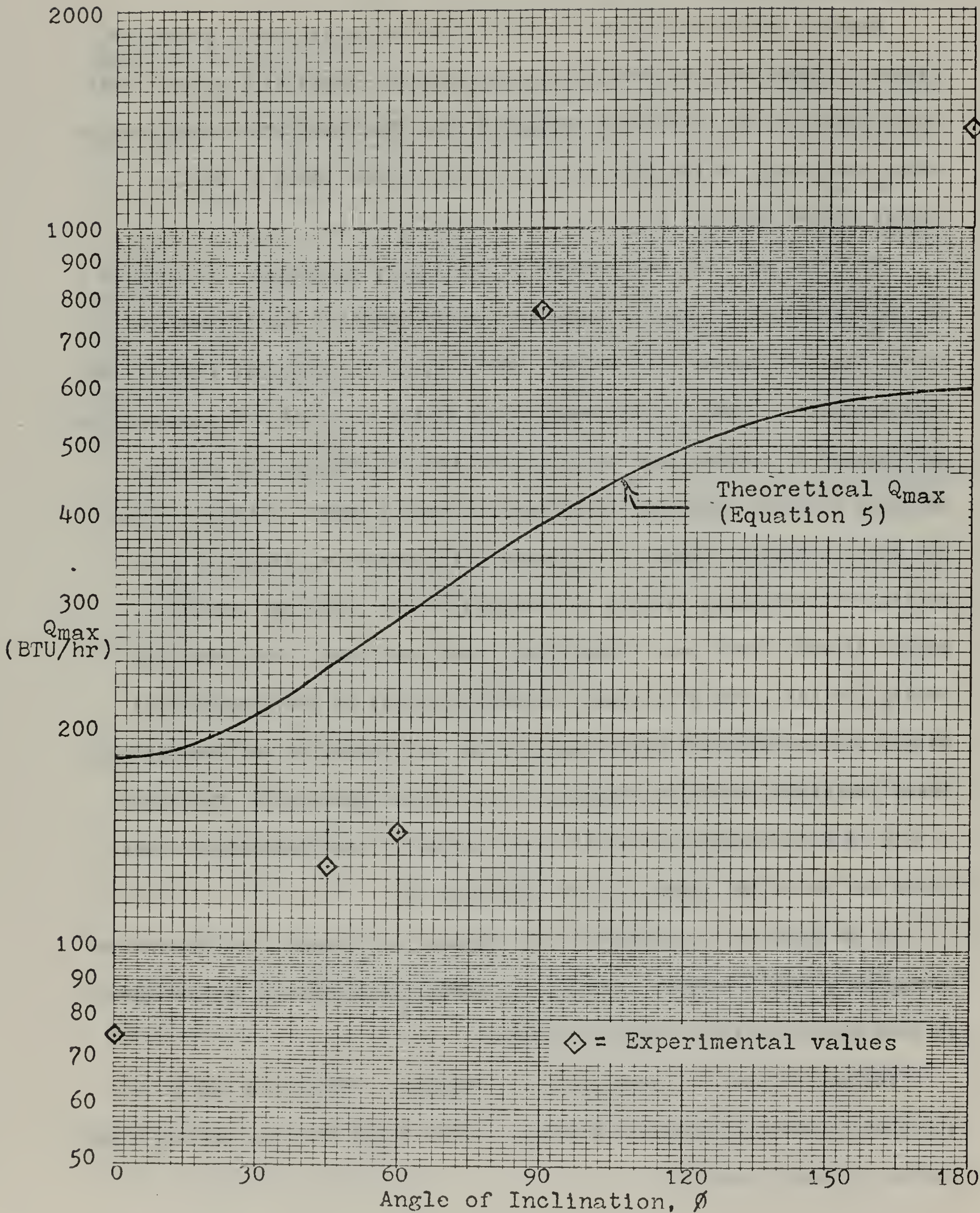


Figure 3. Comparison of Theoretical Calculations and Experimental Results of Maximum Heat Flux vs. Angle of Inclination

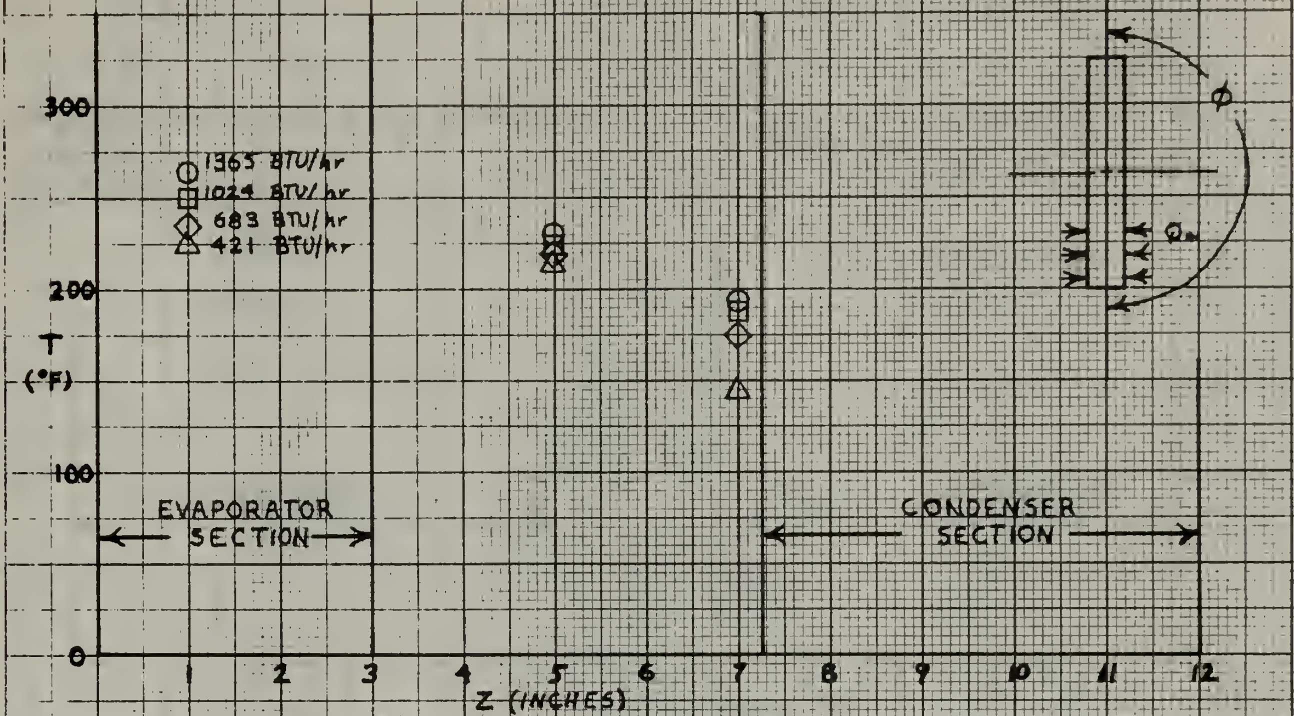


FIGURE 4. AXIAL TEMPERATE DISTRIBUTION
ANGLE OF INCLINATION, $\phi = 180°$

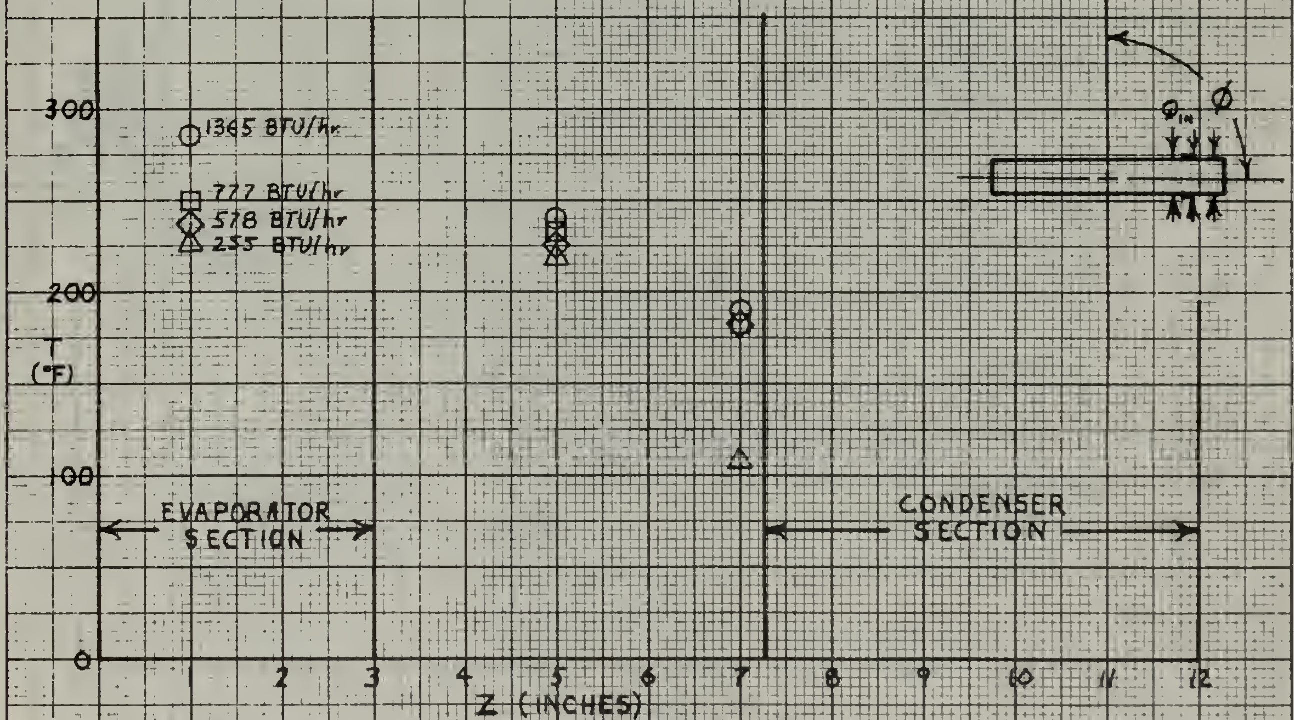


FIGURE 5. AXIAL TEMPERATURE DISTRIBUTION
ANGLE OF INCLINATION, $\phi = 90°$

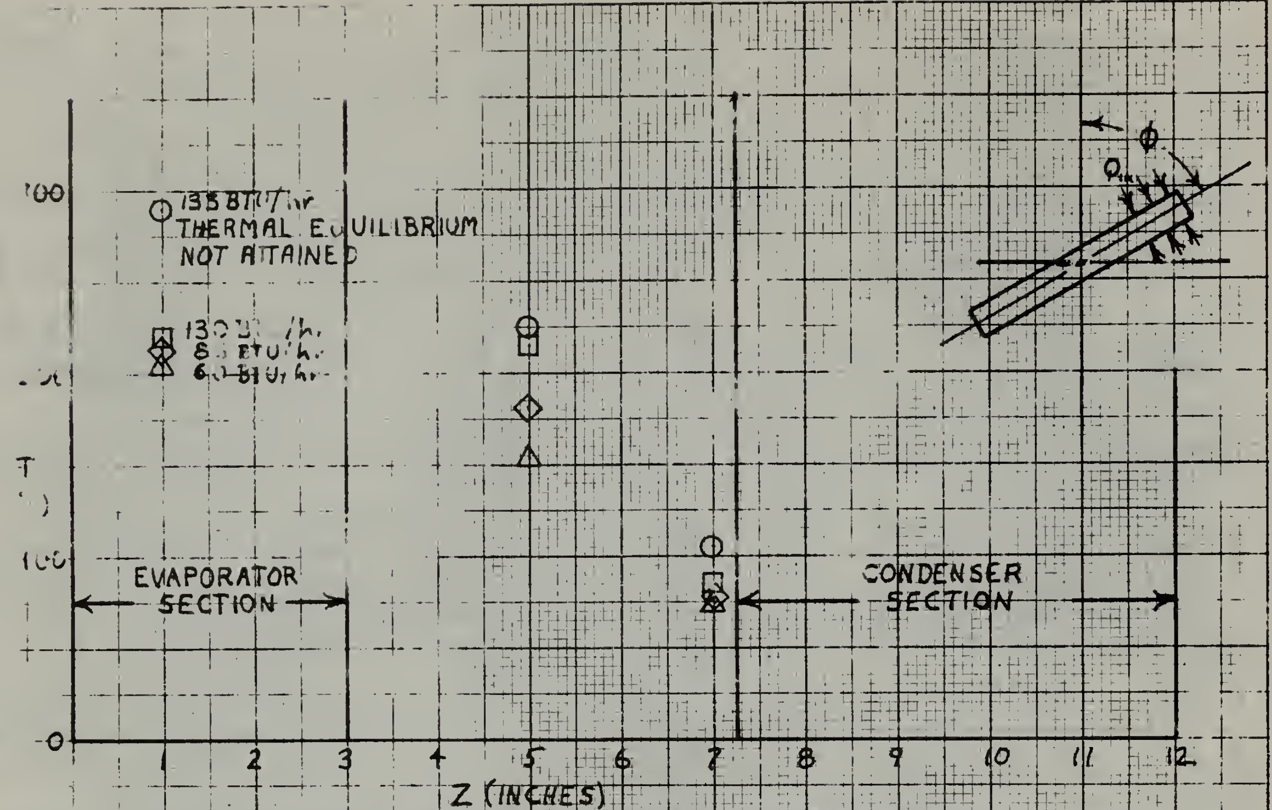


FIGURE 6. AXIAL TEMPERATURE DISTRIBUTION
ANGLE OF INCLINATION, $\phi = 60°$

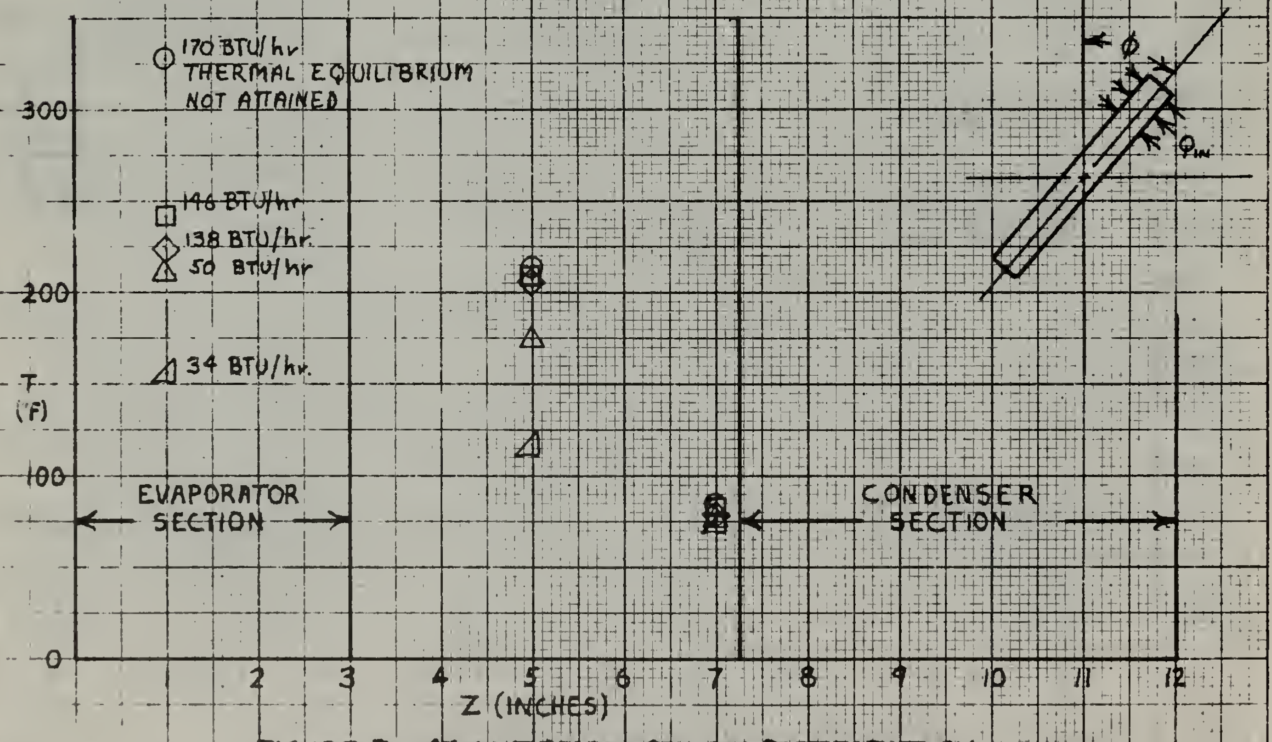


FIGURE 7. AXIAL TEMPERATURE DISTRIBUTION
ANGLE OF INCLINATION, $\phi = 45°$

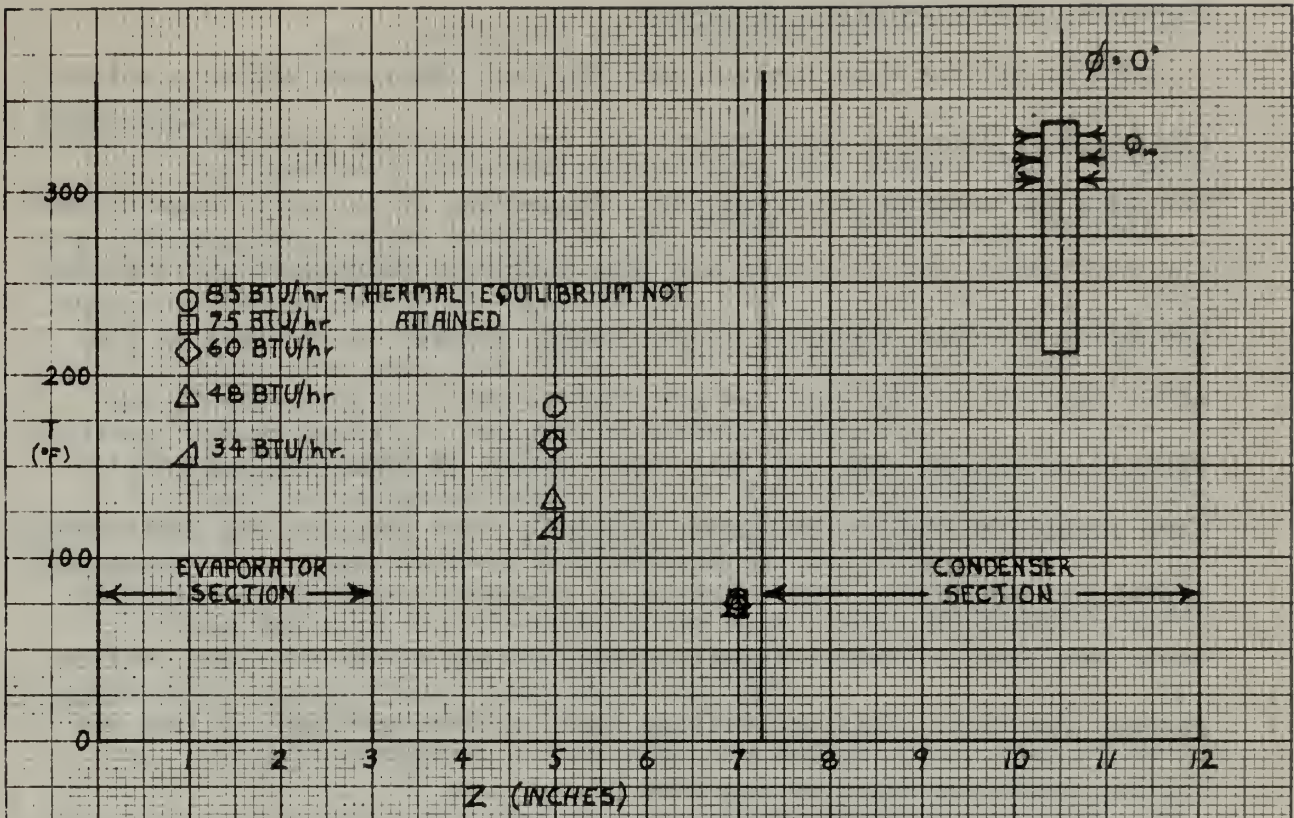


FIGURE 8. AXIAL TEMPERATURE DISTRIBUTION
ANGLE OF INCLINATION, ϕ = 0°

Examination of Pipe After Operation

After all the experimental runs had been completed and data collected, the heat pipe was disassembled and one end of the pipe was cut off. The wick was removed and inspected. Inspection of the wick revealed that the wick and retaining spring had been partially decomposed and oxidized. The cause for this oxidation is not known, however it is entirely possible that all of the acid was not removed from the pipe when it was cleaned the second time and this small amount of remaining acid could have caused the wick to decompose partially when the pipe was subsequently heated and outgassed. The presence of oxidation products in the wick would have had an adverse effect on the wetting of the wick and on flow through the wick. This non-wetting would promote nucleate boiling and boiling noise was heard during some of the runs.

IV. CONCLUSIONS AND RECOMMENDATIONS

Conclusions

1. The results of the experimental work were inadequate to lead to any definite conclusions concerning the effect of gravity or nucleate boiling on the operation of a heat pipe.

2. Great care has to be taken in the manufacture, cleaning, and filling of heat pipes if they are to operate properly.

3. Position of thermocouples must be accurately known in order to accurately determine the heat flux through the evaporator wall.

4. The pressure distribution in the wick cannot be measured by any method that disturbs the integrity of the pipe.

5. Prior to designing a heat pipe, one should obtain all the pertinent information about the wick by experimental means if it is not published or if its validity is in doubt.

Recommendations

1. First, consider the measurement of heat flux into the evaporator. Measurement of power into the heating element does not accurately give the heat flux into the evaporator so some direct means of measurement is required. One method would be to machine small grooves in the outer and inner walls of the evaporator, place thermocouples in the grooves, accurately determine their positions, cover the thermocouples and fill in the grooves with the heat pipe material, and polish the surface so that it is even with the adjacent material. Another method would be to measure the heat flow directly. Geonetics Corporation of San Diego manufactures a device called a Thermal Flux Transducer that could possibly be placed on the evaporator surface between the pipe and the heating element and would measure the heat flux directly.

2. Measurement of axial temperatures in the vapor chamber can be accomplished by inserting a small tube containing the desired number of thermocouples. This could be done with a minimum of difficulty using a CONAX multiple seal gland.

3. Small pressure transducers installed in the wall of the pipe should serve to measure the pressure distribution in the wick while maintaining the integrity of the interior of the pipe.

4. The following method is recommended for determining the properties of a porous wicking material prior to installation in a heat pipe. Three important properties of the wick material are the capillary radius r_c, the permeability factor b, and the screen void factor, ϵ. The value of r_c for a particular wick may be determined by a simple capillary rise experiment. The factors b and ϵ appear only in the term for the pressure drop in the wick and may be grouped as b/ϵ. A value for this term may be determined as follows:

Take a section of wick material of known constant length and cross-sectional area, pass a constant, steady flow of the working fluid through the material and measure the pressure drop. The liquid pressure drop term in equation (5) is

$$\frac{\nu Z}{2\pi(R_w^2 - R^2) \quad r_c^2} \quad \frac{b}{\epsilon} \; \frac{Q}{L}$$

Now $Q/L = \dot{m}$ = mass flow rate,

$2\pi(R_w^2 - R^2)$ represents the cross-sectional area, Z is the length of the sample, ν is known and r_c can be found. Therefore

$$b/\epsilon = \Delta P_1 \quad L/Q \quad \frac{r_c^2 \times \text{area}}{\nu \times \text{length}} \tag{8}$$

5. The cleaning procedure should be thoroughly investigated and tested prior to using.

6. If at all possible, the heat pipe should be designed so that visual observations can be made of the interior of the pipe during operation.

7. The condenser end of the pipe should be flanged so that the pipe can be dismantled for examination after operation or for recleaning if necessary.

8. Build a heat pipe of rectangular cross-section with a wick on either one or three sides. Heat would be added or removed from the sides covered by the wick. The fourth side would be enclosed by either two or three parallel layers of glass, each sheet of glass separated from the other by a small distance and this space between the sheets of glass should be evacuated. Prior to operation, the entire pipe should be uniformly heated at the operating temperature to insure that there will be no condensation on the glass. This device could also be instrumented for pressure and temperature measurement.

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APPENDIX I

Sample Theoretical Calculations

The solution to Cotter's pressure drop equation (5) for maximum theoretical heat flux was determined by the Newton-Raphson method for solving a quadratic equation [12]. The Newton-Raphson method is an iterative technique and in its basic form is given by

$$x_{i+1} = x_i - f(x_i)/f'(x_i)$$

or in terms of Q

$$Q_{next} = Q_1 - f(Q_1)/f'(Q_1)$$

Let $A = (1-4/\pi^2)/(8\rho_v R^4 L^2)$

$$B = (b\nu Z)/(2\pi(R_w^2-R^2)\epsilon r_c^2 L)$$

$$C = \rho_1 g Z \cos\phi/g_o - 2\gamma\cos\theta/r_c$$

Cotter's equation may now be written

$$AQ^2+BQ+C=0$$

Let

$$f(Q) = AQ^2+BQ+C$$

and

$$f'(Q) = 2AQ+B$$

The following calculations are for water and steam at 212°F.

$$R_w = 0.570/12 \text{ ft}$$

$$R = 0.561/12 \text{ ft}$$

assume

$$\cos\phi = 1$$

$$\cos\theta = 1$$

$$b = 20$$

therefore

$$A = .01395 \text{ lb-sec}^2/\text{BTU}^2\text{-ft}^2$$

$$B = 1032.2 \text{ lb-sec/BTU-ft}^2$$

$$C = -52.718 \text{ lb/ft}^2$$

There are two roots, one positive and one negative, to the quadratic equation. The negative root is meaningless, therefore the positive root is the desired solution to the equation. The positive root will be a small positive number, therefore assume a value

$$Q_1 = 0.12$$

as a starting point.

$$Q_{next} = .12 - \frac{71.12}{1032.2} \approx .06$$

Since $\left| \frac{.12-.06}{.12} \right| > .0001$

set $Q_1 = Q_{next} = .06$

and iterate again following the above procedure until

$$\left| \frac{Q_1 - Q_{next}}{Q_1} \right| < .0001$$

The final value of Q_1, the desired root is

$$Q_1 \approx .051 \text{ BTU/sec}$$

APPENDIX II

Error Analysis

One major source of error in determining the maximum capacity of a heat pipe lies in the inaccuracies in determining the heat flux at which operation breaks down. Direct measurement of power into the device by means of a wattmeter or other instrument is inaccurate due to inherent heat losses involved. An attempt was made to measure the heat flux by measuring the temperature at various points in the pipe wall and computing the heat flux by means of the equation for steady state conduction in one dimension. The following error analysis is based on the heat transfer equation for one dimensional, steady state heat transfer in a cylindrical system which is just

$$Q = 2\pi kz(T_o - T_i)/\ln(r_o/r_i)$$

for heat flow into the pipe. The first four terms on the right side of the equation may be grouped into one constant term so that

$$Q = K(T_o - T_i)/\ln(r_o/r_i)$$

Now setting $\Delta T = T_o - T_i$

and taking the derivative

$$\delta Q = \frac{K\,(\delta \Delta T)}{\ln(r_o/r_i)} + \frac{\Delta T\,\delta K}{\ln(r_o/r_i)} - \frac{K\,\Delta T\,\delta(r_o/r_i)}{r_o/r_i(\ln r_o/r_i)^2}$$

$$\frac{\delta Q}{Q} = \frac{\delta \Delta T}{\Delta T} + \frac{\delta K}{K} - \frac{\delta(r_o/r_i)}{(r_o/r_i)\ln(r_o/r_i)}$$

Now K is a constant term and may be assumed to be known accurately with respect to the other terms so that

$$\frac{\delta K}{K} = 0$$

Also, since the desired value for $\delta Q/Q$ is its maximum value, the negative term is replaced by its absolute value; therefore:

$$\frac{\delta Q}{Q} = \frac{\delta \Delta T}{\Delta T} + \frac{\delta (r_o/r_i)}{(r_o/r_i) \ln (r_o/r_i)}$$

Now assume the following values:

$Q = 50$ watts $= 170.65$ BTU/hr

$r_o = 0.720$ inches

$r_i = 0.625$ inches

$K = 2\pi kz = 15.71$ BTU/hr-°F

Therefore:

$$\Delta T = \frac{(170.65)\ln(.720/625)}{15.71} = 1.59°F$$

Now the error in measuring the temperature was due to two causes. The first was the inaccuracies in the digital voltmeter and they were approximately $\pm 2\ \mu$v. The second cause of error arises from the method of calibrating the thermcouples. The thermocouples were calibrated in a steam bath at 212°F and the error was assumed to be constant throughout the temperature range measured. The error in measurement due to the above assumption was estimated to be approximately $\pm 8\ \mu$v. The total temperature error was 8 + 2 or 10 μv. An error of 10 μv is equivalent to a temperature error of approximately 0.4°F for copper-constantan thermocouples. This gives an estimate of

$$\frac{\delta \Delta T}{\Delta T} = \frac{0.4}{1.59} = .25$$

The remaining error was one of thermocouple position error and was based on two premises. The first premise was that the holes for the thermocouples were not true and the second was that the location of the thermocouple junction in the hole was not known.

For purposes of error analysis the assumption was made that the end of the drill bit could deflect one bit diameter between the entrance and the end of the hole, that is, the center of the entrance hole and the center of the bottom of the thermocouple well either are not the same radial distance from the center of the pipe, or have some angular displacement between them, or both. The average size of the thermocouple bead was 0.018 inches in diameter, including insulation. It was assumed that the copper-constantan junction was in the center of the bead. Under the above conditions it would be possible for a given thermocouple to have a maximum error in radial position, Δr, of 0.020+ 0.040-0.018/2=0.051 inches. An error of this magnitude would cause a maximum error in the ratios of two different radii, $\delta (r_o/r_i)$, of 0.169 when r_o and r_i are the values given above.

This gives an estimate of

$$\frac{\delta (r_o/r_i)}{(r_o/r_i)\ln (r_o/r_i)} = \frac{.169}{(1.152)(.1415)} = 1.04$$

Therefore for maximum error $\delta Q/Q$ is:

$$\delta Q/Q \approx 130\%.$$

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DOCUMENT CONTROL DATA - R & D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY *(Corporate author)* Naval Postgraduate School Monterey, California 93940	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP

3. REPORT TITLE

Design and Operation of a Heat Pipe

4. DESCRIPTIVE NOTES *(Type of report and inclusive dates)*
None

5. AUTHOR(S) *(First name, middle initial, last name)*

Burges, Rufus Thurman, Jr.

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
June 1968	44	14

8a. CONTRACT OR GRANT NO. b. PROJECT NO. N/A c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) N/A 9b. OTHER REPORT NO(S) *(Any other numbers that may be assigned this report)*

10. DISTRIBUTION STATEMENT

[redacted]

9/3/70

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY Naval Postgraduate School Monterey, California

13. ABSTRACT

An experimental stainless steel heat pipe using water as the working fluid and 400 mesh stainless steel screen for a wick was designed and tested to determine the effect of gravity and nucleate boiling on heat pipe performance. The results of heat pipe operation at various angles of inclination in a gravity field are presented and compared with the existing theoretical predictions.

The maximum heat flux obtained experimentally at angles of inclination less than 90 degrees was less than the predicted value by a factor of two or three. The maximum heat flux obtained for an angle of inclination greater than 90 degrees was much higher than that predicted. In addition, nucleate boiling noise was detected during operation at angles of inclination greater than 90 degrees.

The experimental results coupled with visual examination of the pipe after operation indicate that the pipe was not performing satisfactorily. Recommendations for a better design of an experimental heat pipe are presented.

DD FORM 1 NOV 65 1473 (PAGE 1)
S/N 0101-807-6811

Unclassified
Security Classification

A-31408

14. KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Heat Pipe						
Heat Transfer						
Gravity						
Nucleate Boiling						
Effective Capillary Radius						
Wick						

[redacted]

[redacted]

[redacted]